HEAT TRANSFER IN FORCED CONVECTION OF FLUID IN POROUS SINTERED METALS

V. M. Eroshenko and L. A. Yaskin

UDC 536.25

The present state of the problem is analyzed and latest results presented for studies of heat transfer during the flow of fluid within porous sintered metals.

The design of internal cooling for porous structural elements subject to the effects of intense surface or internal heat sources primarily demands a knowledge of the relations governing heat transfer between the coolant and the walls inside the porous structure. In the literature, a considerable number of papers [2-14] are devoted to a study of heat transfer within porous sintered metals. As seen below, however, the results obtained can be put to practical use only under known restrictions. Criterial relations for heat transfer which were established in the papers cited are given in Table 1 along with necessary information about the conditions under which experiments were performed, the parameters of the porous samples, and the methods of testing.

In analyzing the experimental data produced, it is expedient to discuss first the technical questions concerning the methods for determination and representation of the volumetric coefficient of heat transfer within pores and the Nusselt number associated with it. In this regard, it is convenient to divide all the studies into two groups.

The work in the first group was based on the use of the computational and experimental method discussed in the general outlines of Grootenhuis and Moore [1]. According to [1], the unknown value of the heat-transfer coefficient α_V can be calculated from solutions of the equations for the internal problem of the temperature distributions in the solid and liquid phases

$$\lambda_w \frac{d^2T_w}{dx^2} = \alpha_v (T_w - T_g) = \rho v c \frac{dT_g}{dx}, \tag{1}$$

if the values of the material and fluid temperatures at the bounding surfaces of the walls are known from experiment. The assumption in [1] concerning the negligibly small value of heat transfer on entrance into a porous wall was used [2] in a realization of this method under experimental conditions. This can lead to an overestimate of the value of the coefficient α_V, as was pointed out in [7].

In a calculation of temperature distributions within a porous wall [3], boundary conditions on entrance into the wall were used which showed little correspondence with ideas about the actual flow pattern and thermal environment in this region [16]. For this reason, the experimental data obtained in [3] for intrapore heat transfer are of insufficient reliability.

Polyaev and Sukhov improved the method of [1] by using a completely physical boundary condition of the second kind for heat transfer between fluid and wall upon entrance [7]. However, values for the thermal conductivities of porous metals given by the expression $\lambda_{wP} = \lambda_{wo}(1 - P)$, and, consequently, overestimated by factors of 1.5-2.5, were used in the analysis of the experimental data in [7].

In the second group of experimental papers belong studies [4-6, 10-14] in which, in analogy with the case of heat transfer in a tube with constant wall temperature, a coefficient of heat transfer within pores was used that was calculated from the mean logarithmic temperature differential

G. M. Krzhizhanovskii Institute of Power Engineering, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 30, No. 1, pp. 5-13, January, 1976. Original article submitted January 29, 1975.

This material is protected by copyright registered in the name of Plenum Publishing Corporation, 227 West 17th Street, New York, N.Y. 10011. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, microfilming, recording or otherwise, without written permission of the publisher. A copy of this article is available from the publisher for $7.50.

$$\alpha_{v_{\log}} = \frac{q_{\text{int}}}{L\,\overline{\Delta T}_{\log}} = \frac{q_{\text{int}}}{L} \cdot \frac{\ln \dfrac{T_{wL} - T_{gL}}{T_{w0} - T_{g0}}}{(T_{wL} - T_{gL}) - (T_{w0} - T_{g0})}. \tag{2}$$

Values of α obtained from Eqs. (1) and (2) can only be identical when there is an insignificant variation of temperature over the thickness of the wall, which is roughly satisfied only in the cooling of thin walls made of highly conducting metals.

Determination of the coefficient of heat transfer within pores from $\Delta T_{\log}$ is coupled with the overcoming of a difficulty which forced the authors to resort to the assumptions given below. We are talking of the determination of the temperature T_{gL} of the fluid on entrance into a pore channel at the internal surface of the wall, which appears in the expression for $\Delta T_{\log}$ and in $q_{\text{int}} = \rho vc(T_{g_0} - T_{gL})$. It does not appear possible to measure this temperature experimentally although attempts have been made to do so [13]. Because of this, the *a priori* assumption was made that heat transfer upon entrance was negligibly small, i.e., $T_{gL} = T_{g\infty}$. Meanwhile, as is well known from [7], heat removal upon entrance into the wall can even exceed internal heat removal in a number of cases.

In [13], they speak of the satisfactory agreement of the values of the coefficient α_V which are obtained as the result of analysis of experimental data in accordance with Eq. (2), on the one hand, and in accordance with the solutions (1) of the internal problem, on the other hand. In the latter case, the somewhat incorrect condition of negligibility of heat transfer upon entrance into the wall was used as a boundary condition. At the present time, the question of the correspondence of $\alpha_{V_{\log}}$ to the value of α_V in Eq. (1) remains open as before.

In concluding the analysis of technical problems, we point out a characteristic feature of all the work — the use of the condition of negligible heat transfer in outflow from pores (see [7], for example).

Despite the seeming abundance of publications on the experimental study of heat transfer within porous sintered metals, one has not succeeded in unifying the published data into a single criterial dependence of the Nusselt number on the controlling parameters

$$\mathrm{Nu}_v = \frac{\alpha_v d^2}{\lambda_g} = f(\mathrm{Re},\ \mathrm{Pr},\ P,\ L/d,\ \lambda_g/\lambda_w,\ T_w/T_g,\ \ldots). \tag{3}$$

This can be explained primarily by the previously indicated differences in the techniques for the determination of the volumetric coefficient of heat transfer within pores (for a pore of a theoretical nature). An important reason for such a situation, which prevents comparison of the results of various workers in a number of cases, is the absence of the required uniformity in the choice of the controlling dimension for Nu_V and Re (Table 1). One can formulate some concept of the nature of the dependence of heat transfer on the controlling parameters appearing in Eq. (3) on the basis of the published data.

Intensification of heat transfer within pores because of an increase in coolant flow rate is expressed in all studies in the form of a power function of the Reynolds number

$$\mathrm{Nu}_v = A\,\mathrm{Re}^n, \tag{4}$$

the nature of which is contradictory to a considerable extent. The value of the exponent n falls in the range from 0.56 [15] to 1.84 [4] according to the data of the different authors, but in most of the work, the quantity n = 0.9-1.3 (Table 1).

There is also great disagreement of results with respect to the dependence of the Nusselt number on the relative wall thickness L/d. Even the first experiments investigating intrapore heat transfer [2] revealed a strong differentiation of experimental points for porous samples differing with respect to wall thickness and particle diameter. However, no one succeeded in establishing some kind of regular relationship in such behavior of the points because of the variation of the parameter L/d. The spread of the data in [2] was explained in [4] as the result of an unfortunate choice of the controlling dimension in Nu_V and Re and it was proposed that the experimental data be analyzed with respect to the characteristic dimension β/α. We point out, however, that the differentiation of data in [2] would be maintained in this case, since the hydraulic coefficients β and α depend mainly on the porosity,

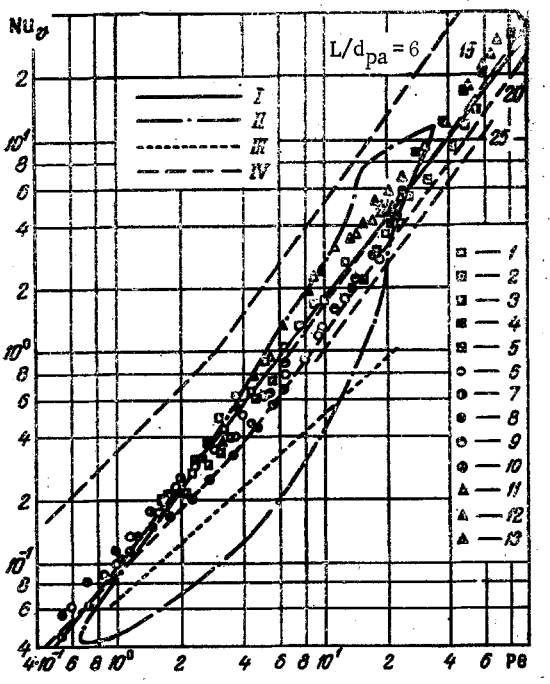

Fig. 1

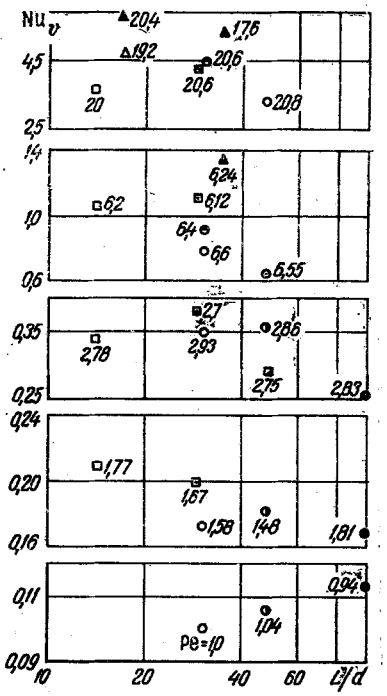

Fig. 2

Fig. 1. Correlation of experimental data on heat transfer during flow of fluid within porous sintered metals: I) from Eq. (9); II) [2]; III) [5]; IV) [10]; 1) stainless steel, P = 0.27, d = 0.35 mm, L/d = 14.5, He; 2) stainless steel, 0.28, 0.35, 29, He; 3) stainless steel, 0.29, 0.35, 48, He; 4) stainless steel, 0.27, 0.35, 14.5, Ar; 5) stainless steel, 0.28, 0.35, 29, Ar; 6) stainless steel, 0.29, 0.1, 30.5, He; 7) stainless steel, 0.31, 0.1, 48, He; 8) stainless steel, 0.31, 0.1, 100, He; 9) stainless steel, 0.29, 0.1, 30.5, Ar; 10) stainless steel, 0.31, 0.1, 48, Ar; 11) bronze, 0.23, 0.3, 17, He; 12) bronze, 0.23, 0.3, 35, He; 13) bronze, 0.23, 0.3, 17, Ar.

Fig. 2. Effect of relative thickness of porous wall on intensity of heat transfer within porous sintered metals.

which remained practically unchanged in the experiments in [2]. It is apparent that the main reason for the inconsistency of the data in [2] as a function of the parameter L/d is the deficiency of the method used for the determination of the coefficient α_v, which employed the somewhat incorrect boundary condition at the internal surface of the wall.

It was established in the experiments in [4] that the Nusselt number for a plate 0.73 mm thick with a porosity of 0.30 was 4.8 times greater than for a plate with L = 4.2 mm and P = 0.52. However, information about the particle sizes in the porous test samples is lacking in that paper.

Studies made in the range L/d_{po} = 50–110 [3] recorded an intensification of heat transfer with increase in relative wall thickness. According to these data, $Nu_v \sim (L/d_{po})^{0.9}$. The directly opposite result, $Nu_v \sim d_{pa}/L$, was obtained in the experiments in [10] for porous plates with relative thicknesses L/d_{pa} = 6, 15, 20, and 25.

TABLE 1. Review of Basic Experimental Work on Heat Transfer in Porous Sintered Metals

Reference, year	Experimental condition	Method of determination of α_V	Parameters of porous samples (dimensions d, L in mm)
Grootenhuis et al. [2], 1951	Radiation heating	From solution of internal problem; $T_{gL} = T_{g\infty}$	Bronze; d_{pa} = 0.049-0.401; L/d_{pc} = 9.0-56; P = 0.333-0.395
Bernicker [3], 1960	"		Stainless steel; d_{po} = 0.013-0.0465; L = 0.79-3.18; P = 0.35-0.58
Druzhinin [4], 1961	Induction heating	From ΔT_{log} with $T_{gL} = T_{g\infty}$	Stainless steel; 1) P = 0.30, L = 0.73; 2) P = 0.52, L = 4.2; d not given
Kharchenko [5], 1968	Nonstationary cooling	From $\Delta T = (T_{wL} - T_{g\infty})/2$	Copper; d_{pa} = 0.1, 0.15; L = 3; P = 0.35
Stradomskii et al. [6], 1968	Induction heating	From ΔT_{log} with $T_{gL} = T_{g\infty}$	Nickel wool; d = 0.25; P = 0.456, 0.374
Polyaev, Sukhov [7], 1969	Ohmic heating	From solution of internal problem	Nichrome; d_{pa} = 0.02-0.05; L = 0.5, 1.4, 5.0; P = 0.25, 0.32, 0.4
Bayley, Turner [9], 1970	"	Not given	Bronze, stainless steel, alloys; d_{po} = 0.001-0.11; L/d_{po} = 10-1630; P = 0.144-0.648; d_{pa} not given
Maksimov [10], 1970	Induction and ohmic heating	From ΔT_{log} with $T_{gL} = T_{g\infty}$	Stainless steel, nichrome, nickel; d_{pa} = 0.065-0.33; L = 1.3-8.15; P = 0.31-0.556
Maksimov et al. [11], 1970	Induction heating	"	"
Maksimov, Stradomskii [12], 1971	"	"	Stainless steel; d_{pa} = 0.065; L = 1.3; L/d_{pa} = 20; P = 0.3
Maksimov, Stradomskii [13], 1972	Induction and ohmic heating	"	"
Boiko et al. [14], 1972	Radiation heating	"	Stainless steel; d_{pa} = 0.3; L = 4.75; P = 0.333

Analysis of the data from [3, 10] should be discussed. If the correlation of Nu_V and L/d actually occurs in the form of a direct [3] or inverse [10] proportional relationship, this must mean that the intensity of heat transfer has to increase [3] or decrease [10] over the wall thickness in the direction of the fluid motion. This places in doubt the traditional condition of constancy of the heat-transfer coefficient over wall thickness, $\alpha_V(x)$ = const, used in all calculations of porous cooling.

The experimental data of [9] in the range L/d_{po} = 10-29 has a certain spread (about 30-100%) which is impossible to explain through the effect of the parameter L/d_{po}. At higher ratios L/d_{po} = 79 and 680, heat transfer deteriorates by factors of 2.5-4, respectively. The dependence of the Nusselt number on the parameter L/d was not explained in [7].

The effect of coolant properties on heat transfer within pores was investigated in [12] over a broad range of values of the Prandtl number, Pr = 0.7-140. As was shown, the Prandtl number should appear in the criterial equation (4) for heat transfer to the same power as the Reynolds number.

A noticeable effect on heat transfer by such factors as porosity of the metal, shape of particles in the original powder, ratio of the coefficients of thermal conductivity of the phases, or method of heating the porous sample was not observed in the experiments in [10].

Coolant	Dimension	Re	Nu_v	Range of Re	Criterial heat-transfer equation
Air	d_{pa}	$\frac{Gd_{pa}}{F\mu P}$	$\frac{\alpha_v d_{pa}^2}{6(1-P)\lambda}$	2-160	
Air	d_{po}	$\frac{GL}{F\mu}$	$\frac{\alpha_v d_{po}^2}{\lambda}$	14-600	$Nu_v = 0.0014Re^{0.9}$
Air	β/α	$\frac{G(\beta/\alpha)}{F\mu}$	$\frac{\alpha_v(\beta/\alpha)^2}{\lambda}$	0.014-0.14 0.014-0.07	$Nu_v = 0.0286Re^{1.84}$ $Nu_v = 0.0060Re^{1.84}$
Air	d_{pa}	$\frac{Gd_{pa}}{F\mu P}$	$\frac{\alpha_v d_{pa}^2}{6(1-P)\lambda}$	3.5-90	$Nu_v = 0.042Re^{0.9}$
Air	β/α	$\frac{G(\beta/\alpha)}{F\mu}$	$\frac{\alpha_v(\beta/\alpha)^2}{\lambda}$	0.04-10	$Nu_v = 0.000425Re^{1.2}$
Air	d_{po}	$\frac{Gd_{po}}{F\mu P}$	$\frac{\alpha_v d_{po}^2}{\lambda}$	0.4-5	$Nu_v = 0.28Re^{1.2}P^{2.9}$
Air	d_{po}	$\frac{Gd_{po}}{F\mu P}$	$\frac{\alpha_v d_{po}^2}{6(1-P)\lambda}$	1-45	$Nu_v \sim Re$
Air Nitrogen Helium	d_{pa}	$\frac{Gd_{pa}}{F\mu}$	$\frac{\alpha_v d_{po}^2}{\lambda}$	0.2-5 5-250	$Nu_v = 1.25Red_{pa}/L$ $Nu_v = 0.8Re^{1.3}d_{pa}/L$
Air	β/α	$\frac{G(\beta/\alpha)}{F\mu}$	$\frac{\alpha_v(\beta/\alpha)^2}{\lambda}$	0.02-1.6	$Nu_v = 0.038Re^{1.34}d_{pa}/L$
Air, nitrogen, alcohol, oil	"	"	"	Pr = 0.7-140, 0.003-0.3	$Nu_v = 0.005Pe$
Air	d_{pa}	$\frac{Gd_{pa}}{F\mu}$	$\frac{\alpha_v d_{pa}^2}{\lambda}$	0.27-18	$Nu_v = 0.08Re$
Helium, argon	d_{pa}	$\frac{Gd_{pa}}{F\mu P}$	$\frac{\alpha_v d_{pa}^2}{6(1-P)\lambda}$	3-120	$Nu_v = 0.0175Pe$

The dependence of heat transfer on porosity of the metal established in [3, 7] is of a contradictory nature, as is easily shown (Table 1).

The value of the temperature factor $T_w/\bar{T}_g$ was close to one in all studies.

The analysis made of the state of the investigations of heat transfer during flow of a fluid within porous sintered metals shows that the methods for determination of the coefficient α_v in most cases, and in part the results obtained, give rise to a number of questions, and occasionally objections, with the basic established correlations being marked by extreme inconsistency. For this reason, the known experimental data should be considered to be preliminary. They only give a provisional representation of the actual values of the coefficient of intrapore heat transfer.

Plotted in Fig. 1 are existing published results of experiments on heat transfer within porous sintered metals correlated with respect to the diameter of the particles of the material. As is clear from the figure, the spread of the experimental data of the various authors is quite considerable and may amount to an order of magnitude. It is apparent that such a situation hardly produces any particularly regular behavior of heat-transfer processes within porous sintered metals of identical structure. The reason for this should be sought in the fundamental difference between the various methods for determining the coefficient of heat transfer within pores. All this calls for further studies in this essentially poorly studied

field of heat transfer. Primary consideration should be given to ensuring the correctness of the method used for the determination of the coefficient of heat transfer within pores.

In addition to an analysis of the methods and results of known work, the purpose of the present study is a refinement of the nature of the criterial dependence of heat transfer on Reynolds number and relative wall thickness for forced convection of a fluid in porous sintered metals.

The foundation for the method developed for the computational and experimental determination of α_v, the coefficient of heat transfer within pores, was an idea in [1] which was extended by us to radiative heating of porous samples. A solution of Eq. (1) for the internal problem of porous cooling was obtained [14] for the temperature of the outer surface of a porous wall subjected predominantly to the effects of thermal radiation,

$$T_{w0} = T_{g\infty} + 0.5\,\frac{Q_w}{Gc}\left(\sqrt{1+\frac{\mathrm{Pe}^2}{\mathrm{Nu}_v}\cdot\frac{\lambda_g}{\lambda_w}}+1\right). \tag{5}$$

The limits of applicability of Eq. (5) are established by the condition

$$L/d \gtrsim 4\,\mathrm{Pe}/\mathrm{Nu}_v\sqrt{1+\frac{\mathrm{Pe}^2}{\mathrm{Nu}_v}\cdot\frac{\lambda_g}{\lambda_w}}, \tag{6}$$

which is satisfied for most modes of porous cooling.

Equation (5) can be written explicitly for Nu_v in the form

$$\mathrm{Nu}_v = \frac{\mathrm{Pe}^2\,\lambda_g/\lambda_w}{\theta_{w0}(\theta_{w0}-1)}, \tag{7}$$

where $\theta_{w0} = Gc(T_{w0} - T_{g\infty})/Q_w$.

If the wall temperature is measured experimentally not at the outer surface but at a section with the coordinate x_1, a conversion of the temperature T_{w1} into the temperature T_{w0} is made before analysis of experimental data in accordance with Eq. (7):

$$\theta_{w0}\exp\left(-\frac{1}{\theta_{w0}}\cdot\frac{\rho v c x_1}{\lambda_w}\right) = Gc(T_{w1}-T_{g\infty})/Q_w. \tag{8}$$

Equation (8) is a solution of the internal problem for the temperature distribution in a wall with a thickness satisfying the condition (6).

It is then clear that it is sufficient to have experimentally measured values of the wall temperatures T_{w0} (or T_{w1}), of the coolant temperature $T_{g\infty}$ ahead of the wall, and also of the absorbed radiation flux Q_w in order to calculate Nu_v corresponding to the intensity of intrapore heat transfer from the simple equations (7) and (8).

The method of investigation described was realized in experiments performed with a radiative heating device [14]. Porous test samples were prepared by powder metallurgy from powders of Kh18N9T steel and OF 10-1 bronze with particles that were nearly spherical. The quantities Q_w and λ appearing in Eqs. (7) and (8) were measured by a standard procedure of the first kind [14]. Argon and helium were used as coolants. A criterial analysis of experimental data for heat transfer was carried out with respect to the diameter of the original particles of the material. The studies covered the following ranges of controlling parameters: $\mathrm{Pe} = 5.6\cdot10^{-1}$–$0.95\cdot10^{2}$; Pr = 0.66 (Ar), 0.68 (He); $L/d_{pa} = 14.5$–100; $\lambda_g/\lambda_w = 1.3\cdot10^{-3}$–$2.7\cdot10^{-2}$; P = 0.23–0.31.

Experimental data for the intensity of heat transfer within porous walls of varying relative thickness are shown in Fig. 2. It is clear that within the limits of the scatter of experimental data for a fixed value of L/d_{pa}, there is not observed a dependence of Nu_v on the geometric factor L/d_{pa}, which varies by almost an order of magnitude in the experiments. This conclusion may be the basis for the condition α_v = const used in calculations of porous cooling. The factors λ_g/λ_w and P had no effect on heat transfer in the experiments.

A correlation of the results of 13 sets of experiments is shown in Fig. 1. Within ±35%, the experimental data are approximated by a single generalizing dependence on the Peclet number,

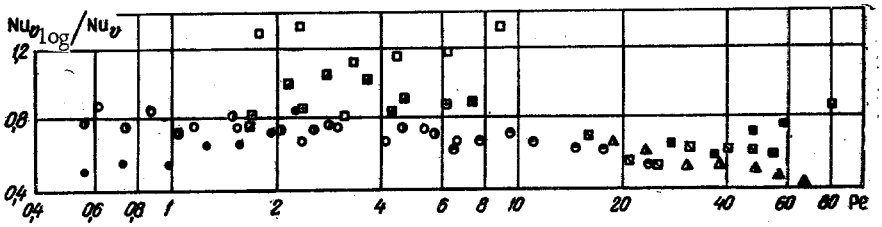

Fig. 3. Comparison of values of Nusselt numbers calculated from Eq. (7) — Nu_V — and from Eq. (2) — Nu_V log.

$$Nu_v = 0.1\ Pe^{1.25}. \tag{9}$$

The data in Fig. 3 demonstrate that the determination of the coefficient α_V from solutions of the internal problem (7) and (8) and from Eq. (2) leads to agreement of the quantities α_V and $\alpha_{V_{log}}$ in particular cases, although an underestimate of the actual value of the coefficient α_V by as much as 50% is also possible.

Considering the basic results of the experiments, we turn our attention to the high value of the power of the Peclet number (n = 1.25) in Eq. (9) and also to the experimental fact that in the range of small Peclet numbers Pe 1, Nu_V does not have its lower asymptotic level and can take on values much less than one. The observed behavior of heat transfer within pores hardly can belong to the class of traditional hypotheses in the theory of convective heat transfer for viscous stabilized flow of continuous media in smooth tubes or for flow around isolated spherical bodies. Therefore, a physical interpretation is required for the mechanism of intrapore heat transfer. It appears rather reasonable to explain the features of heat transfer within porous metals through the effects of the following possible transport processes: 1) viscous molecular flow modes in the narrowest pore passages (Knudsen effect); 2) detached flows within pores with formation of stagnation regions; 3) heat transport by axial thermal conductivity in the fluid in the region of small Peclet numbers Pe $\lesssim$ 1-10.

NOTATION

T, temperature; x, coordinate along wall thickness; Q, thermal flux; G, flow rate; ρ, density; c, isobaric heat capacity of fluid; λ, coefficient of thermal conductivity; F, cross-sectional area; d_{pa}, particle diameter; d_{po}, pore diameter; L, wall thickness; α_V, volumetric coefficient of heat transfer within pores; β/α, ratio of inertial and viscous coefficients of hydraulic resistance in the modified Darcy equation; $Nu_V = \alpha_V d_{pa}^2/\lambda_g$, Nusselt number; $Pe = Gd_{pa}c/F\lambda_g$, Peclet number; Pr, Prandtl number; $\theta_W = (T_W - T_{g\infty})/(T_{go} - T_{g\infty})$, dimensionless wall temperature; $T_{go} = T_{g\infty} + Q/Gc$, fluid temperature when emerging from wall; P, porosity. Indices: w, wall; g, fluid; 0, heat-absorbing surface of wall; L, surface of wall where fluid enters; ∞, region ahead of wall.

LITERATURE CITED

1. P. Grootenhuis and N. P. L. Moore, Proceedings of the Seventh International Congress for Applied Mechanics, Vol. 3 (1948), p. 106.
2. P. Grootenhuis, R. C. A. Mackworth, and O. A. Saunders, General Discussion on Heat Transfer, London (1951), p. 363.
3. R. P. Bernicker, ASME Paper No. 60-WA-223, Aug., 1960.
4. S. A. Druzhinin, Teploénergetika, No. 9 (1961).
5. V. N. Kharchenko, Inzh.-Fiz., Zh., 15, No. 1 (1968).
6. M. V. Stradomskii, E. A. Maksimov, and A. G. Kastornov, Heat and Mass Transfer, Vol. 1 [in Russian], Moscow (1968), p. 72.
7. V. M. Polyaev and A. V. Sukhov, Izv. Vyssh. Uchebn. Zaved., Mashinostr., No. 8 (1969).
8. F. J. Bayley and A. B. Turner, The Aeronautical Journal, 72, 696 (1968).
9. Bayley and Turner, Énerget. Mashiny Ustanovki, 92, No. 4 (1970).
10. E. A. Maksimov, Author's Abstract of Candidate's Dissertation (1970).
11. E. A. Maksimov, V. S. Pugin, and M. V. Stradomskii, Teplofiz. Teplotekh., No. 17 (1970).
12. E. A. Maksimov and M. V. Stradomskii, Inzh.-Fiz. Zh., 20, No. 4 (1971).
13. E. A. Maksimov and M. V. Stradomskii, Teplofiz. Teplotekh., No. 22, 1972).
14. A. N. Boiko, V. M. Eroshenko, V. P. Motulevich, and L. A. Yaskin, Inzh.-Fiz. Zh., 23,

No. 5 (1972).
15. L. Green, ASME Paper 57-HT-19, University of Pennsylvania (1957).
16. D. A. Nealy and P. W. McFadden, J. Heat Transfer, 91, 284 (1969).

INVESTIGATION OF HEAT TRANSFER IN THE SPACE ABOVE AN INHOMOGENEOUS FLUIDIZED BED

A. P. Baskakov and S. V. Zvyagin

UDC 621.783.2:66.096.5

Heat transfer from the rarefied to the dense phase of a fluidized bed by feeding a stream of hot air to the rarefied phase (space above the bed) was investigated.

In a number of studies [1, 2] devoted to the flow of horizontal streams (isothermal and nonisothermal) into a fluidized bed the length of the stream was found and the problem of heat transfer from the jet into the fluidized bed was solved. However, for some fluidized bed processes there is a need for a marked increase of the range of the horizontal streams, which is attained upon outflow of the stream into the space above the bed [3].

Investigations of the structure of the above-bed space [4-6] show that there are two zones with a different character of the distribution of the solid phase: a lower ejection (splash) zone and an upper fluidized bed (pneumatic transport) zone. The relationship of the heights of these zones depends on the size of the particles of the fine-grained material and fluidization regime. Some theoretical considerations on the structure of the zones are given in [4]. In the splash zone the average concentration of particles drops steeply with respect to height and the relative fluctuations of the density are high. In the pneumatic transport zone the average concentration of particles is 2-3 orders lower than in the main bed, and the relative fluctuations are small and are determined mainly by the degree of turbulence of the flow.

Investigations of the above-bed space in apparatuses of various sizes by the filtration method [5], cutoff method [6], and also by a capacitance-type transducer and the x-ray method [4] showed that in semilogarithmic coordinates the height distribution of the concentration (at a constant velocity of the gas flow) is described by two straight lines. The first characterizes the marked drop of the concentration of particles with height in the splash zone and corresponds to the region of inertial movement of the particles. The second more sloping line corresponds to a decrease of the concentration of particles with height in the pneumatic transport zone. The point of inflection of the lines of concentration with height corresponds to the boundary between the splash and pneumatic transport zone.

The decrease of the particle concentration in the splash zone is easily explained [4]: the greater the height, the fewer particles thrown up to this height. The causes of the decrease of particle concentration over the height of the pneumatic transport zone are not discussed in the literature. In the case of a gas velocity constant over the cross section and in time there should be no particles at all in this zone, especially particles circulating between this zone and the fluidized bed, since the free-fall velocity of the particles used is greater than the average gas velocity. The presence of particles in this zone [6] and the sufficiently large coefficients of heat transfer between it and the core of the fluidized bed apparently are the consequence of gas circulation in the above-bed space, as a result of which the local and instantaneous gas velocities can differ considerably from the average not only in magnitude but also in sign. Actually, it is difficult to imagine that at a velocity of

S. M. Kirov Ural Polytechnic Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 30, No. 1, pp. 14-17, January, 1976. Original article submitted February 12, 1975.

This material is protected by copyright registered in the name of Plenum Publishing Corporation, 227 West 17th Street, New York, N.Y. 10011. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, microfilming, recording or otherwise, without written permission of the publisher. A copy of this article is available from the publisher for $7.50.